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Fracture Analysis of the High-Frequency Sail Array Projector's Titanium Mounting Bolts Subjected to UNDEX Shock

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Naval Undersea Warfare Center Division Newport, Rhode Island

PREFACE

This report was conducted under Project No. CS0111, "High-Frequency Sail Array" program, principal investigator H. David Jones (Code 15233). The sponsoring activity is the Naval Sea Systems Command, program manager Julie C. White (PMS-450E2).

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The installation of the High-Frequency Sail Array (HFSA) projector is subject to tight alignment tolerances. Any installation modifications necessitate the use of a machined shim positioned within the projector housing-to-ship foundation joint. Because this shim increases the stresses within the titanium mounting bolts, subsequent fractures must be evaluated for survivability.

A linear elastic fracture analysis was performed on the HFSA mounting bolts for the following reasons: (1) to predict the critical crack size, (2) to ensure that crack growth for a specified shock loading case would not occur, and (3) to determine the minimum mode-I fracture toughness required for the mounting bolts.

Electric Boat Corporation's global shock model of the projector, called the Dynamic Design Analysis Method, was used to predict the maximum bolt shear stress for a specific underwater explosion. Localized bolt bending stresses caused by joint eccentricity were then determined using a closed-form solution. After the relevant stress data were determined, the NASGRO Fracture and Fatigue Prediction Program was invoked to assess failure by fracture. The mode-I stress intensity factor was computed and compared to the fracture toughness of the recommended bolt material.

14. SUBJECT TERMS

Axial Stress   Bending Stress   Bolt Shear Stress   Fracture Toughness
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LIST OF SYMBOLS

a	Flaw depth (in.)
a_{cr}	Critical flaw size
A_s	Cross-sectional area of bolt shank region (in.^{2})
A_t	Tensile cross-sectional area of bolt threaded region (in.^{2})
c	Flaw half-width (in.)
d_s	Diameter of bolt shank region (in.)
d_t	Tensile diameter of bolt (in.)
$F_0(\lambda)$	Axial stress magnification factor
$F_1(\lambda)$	Bending stress magnification factor
F_{pre}	Bolt pre-load force (kip)
g	Acceleration due to gravity (386.4 in./s^2)
κ	Torque coefficient
K_1	Mode-I stress intensity factor (ksi in.$^{1/2}$)
K_{1c}	Mode-I plane strain fracture toughness (ksi in.$^{1/2}$)
L	Moment arm for bolt bending (in.)
λ	Ratio of flaw depth to bolt diameter
P	Bolt shear force (kip)
r_s	Radius of bolt shank region (in.)
S_0	Pre-load stress in bolt shank region
S_1	Bending stress in bolt shank region (ksi)
S_{pre}	Pre-load stress in bolt threaded region (ksi)
S_{total}	Combined tensile stress, sum of S_0 and S_1 (ksi)
S_{yield}	Tensile yield stress of bolt (ksi)
τ_b	Averaged bolt shear stress computed by DDAM (ksi)
t_{found}	Thickness of ship foundation (in.)
t_{s_g10}	Thickness of G10 shim (in.)
t_{s_cres}	Thickness of CRES shim (in.)
t_{tab}	Thickness of projector tab (in.)
T	Torque (ft-lb)

LIST OF ABBREVIATIONS AND ACRONYMS

CRES	Corrosion-resistant stainless steel
DDAM	Dynamic Design Analysis Method
EB	Electric Boat Corporation
HFSA	High-Frequency Sail Array
ICD	Installation control drawing
kip	Kilopound
ksi	kilopounds per square inch
LWWAA	Lightweight wide aperture array
NAVSEA	Naval Sea Systems Command
SIF	Stress intensity factor
UNC	Unified coarse
UNDEX	Underwater explosion

FRACTURE ANALYSIS OF THE HIGH-FREQUENCY SAIL ARRAY PROJECTOR'S TITANIUM MOUNTING BOLTS SUBJECTED TO UNDEX SHOCK

1. INTRODUCTION

The installation of the High-Frequency Sail Array (HFSA) projector is subject to tight alignment tolerances. Any installation modifications necessitate the use of a machined shim positioned within the projector housing-to-ship foundation joint. The presence of this shim, however, increases the load path eccentricity and resulting stresses within the titanium mounting bolts; therefore, subsequent fractures must be evaluated for survivability.

A linear elastic fracture analysis was performed on the HFSA mounting bolts to

1. predict the critical crack size,
2. ensure that crack growth for a specified shock loading case would not occur, and
3. determine the minimum mode-I fracture toughness required for the mounting bolts.

Electric Boat (EB) Corporation's global shock model of the projector, called the Dynamic Design Analysis Method (DDAM),[1] was used to predict the maximum bolt shear stress for a specific underwater explosion (UNDEX). Localized bolt bending stresses caused by joint eccentricity were then determined using a closed-form solution (see section 3).

After the relevant stresses were determined, the NASGRO Fracture and Fatigue Prediction Program[2] was invoked to assess failure by fracture. The mode-I stress intensity factor (SIF) K_1 was computed and compared to the fracture toughness K_{1c} of the recommended bolt material (see section 4).

2. PROJECTOR JOINT AND MATERIAL DESCRIPTIONS

Naval Sea Systems Command (NAVSEA) installation control drawing (ICD) 5612775, revision G,[3] describes the HFSA projector (figure 1) and the projector tabs-to-ship foundation joints (figure 2).

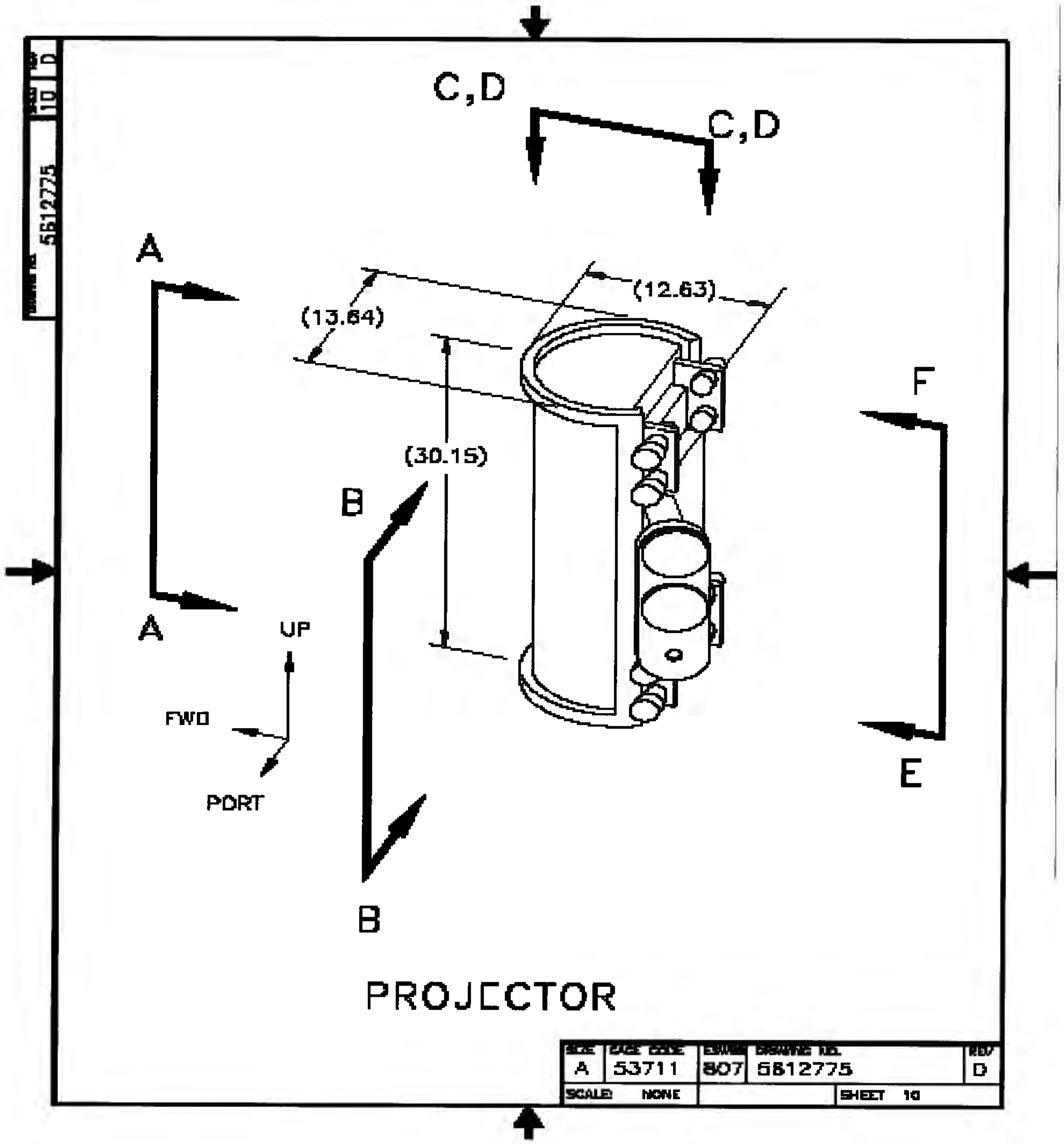

Figure 1. Assembly View of the High-Frequency Sail Array Projector

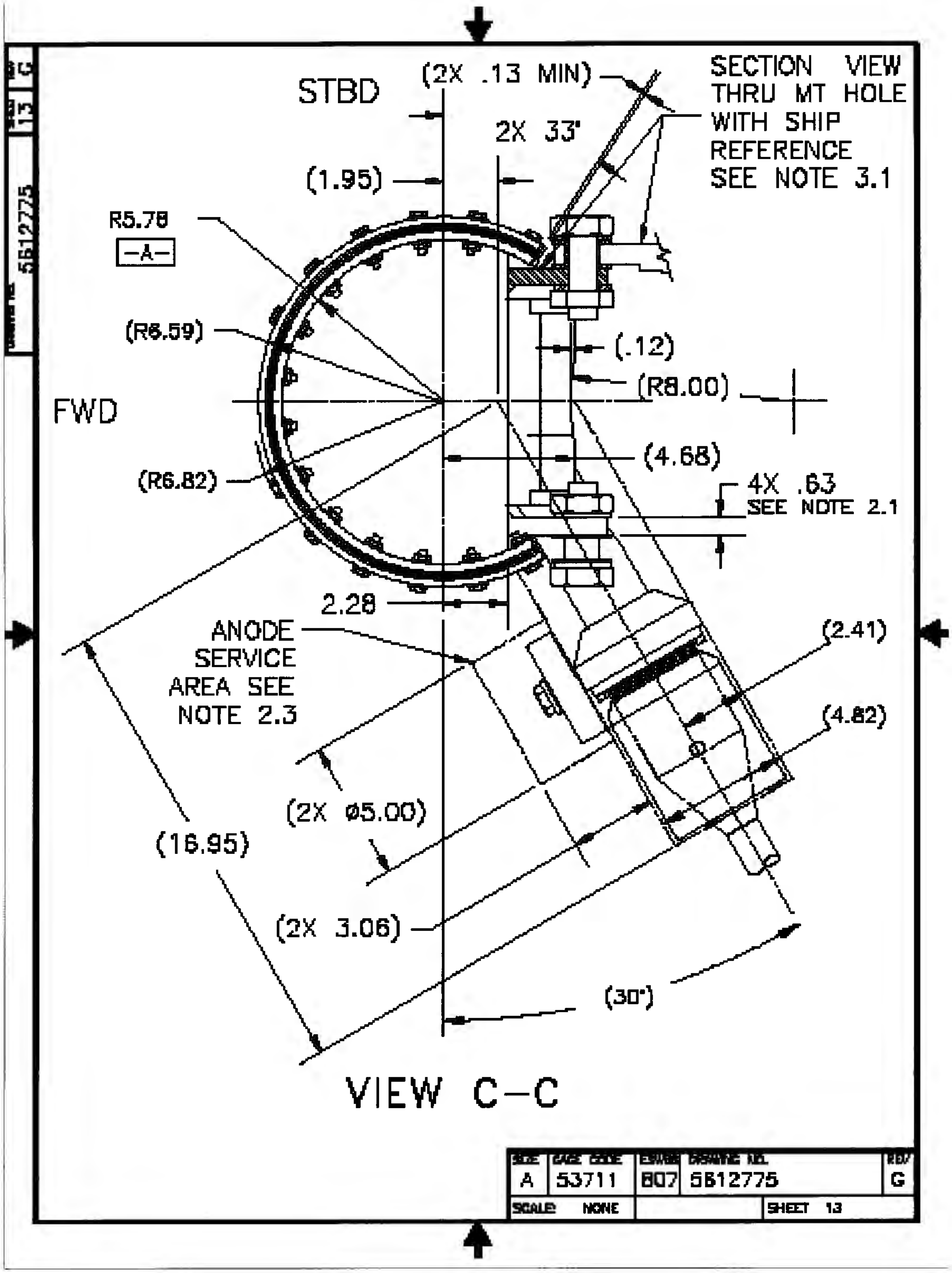

Figure 2. Sectional View Showing Projector Tabs-to-Ship Foundation Joints

The joint results in the eccentric, single-shear loading arrangement as shown in figure 3. Four titanium tabs fasten the projector housing to the ship foundation. Each tab has two 1-inch-diameter clearance holes to accommodate placement of 1-inch-diameter titanium mounting bolts.

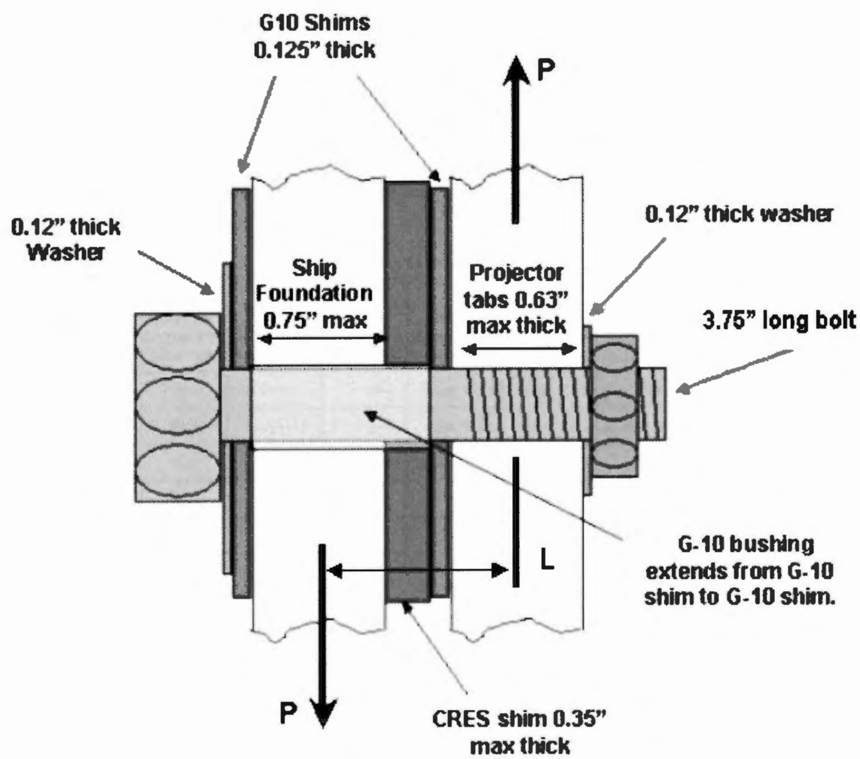

Figure 3. Details of Projector Tab-to-Ship Foundation Bolted Joint

The mounting bolts are hexhead, re-crystallization-annealed titanium T7 with a tensile yield stress S_{yield} of 115.00 ksi in accordance with military specification MIL-S-1222[4] (see table 1). Bolt threads are unified-coarse (UNC) rolled. The tensile diameter d_t and shank diameter d_s are 0.878 inch and 1.000 inch, respectively. The overall bolt length is 3.750 inches. Two 0.120-inch-thick titanium washers are used as shown in figure 3.

The ship foundation has a local thickness of 0.750 inch in the vicinity of where the projector tabs attach. The projector tabs are 0.630 inch thick. As shown in figure 3, two shims separate these components: a corrosion-resistant stainless steel (CRES) shim (selected by EB) that has a maximum thickness of 0.350 inch and a glass-reinforced plastic G10 shim that is 0.125 inch thick. A second G10 shim is positioned between the washer at the bolt head end and the ship foundation. A G10 bushing is included in each hole of the projector tab-to-ship foundation joint. The G10 shims and bushings are used to provide galvanic isolation between the projector bolts and ship foundation.

Table 1. Material Properties of the Titanium T7 Alloy (from MIL-S-1222[4])

				Full-size fasteners				Machined specimens from fasteners or on parent stock			
	Grade	Heat treatment or condition 1/	Nominal diameter (inches)	Tensile strength ksi	Yield strength ksi min	Proof stress ksi min	Rockwell hardness	Tensile strength ksi min	Yield strength ksi min	Elongation in 4D percent min	Reduction of area percent min
Titanium alloy	T7	Annealed	1/4 to 1-1/2	125 min	115 min	----	----	125	115	12	20
			Over 1-1/2 to 2	120 min	110 min	----	----	120	110	12	20
			Over 2	115 min	105 min	----	----	115	105	12	20
		Solution treated and aged	1/4 to 1	145 min	135	----	----	145	135	12	20
			Over 1	135 min	125	----	----	135	125	12	20

3. STRESS ANALYSIS OF SHOCK LOAD CASE

The eccentric load path between the individual projector tabs and ship foundation results in the generation of highly localized bending stresses along the bolts. Three shock cases were examined using DDAM for UNDEX events oriented fore/aft, port and starboard athwartship, and vertical to the submarine.

As predicted by EB's DDAM analysis,[5] the port athwartship UNDEX event produced the maximum projector bolt shear stress τ_b of 30.60 ksi. The port athwartship acceleration exerted on the projector was 175 g (1 g = 386.4 in./s^2). The weight of the projector assembly is 400 pounds in air and 300 pounds when it is submerged in sea water. The DDAM analysis was not conducted to predict bending stresses in the titanium bolts; rather, it was conducted to predict the global reaction forces of the projector assembly only. With the peak dynamic shear stress having been established, a simplified closed-form solution was used to predict the localized bolt bending stress.

Localized bending stresses on the titanium projector mounting bolts, resulting from the eccentric load path between the ship foundation and the projector tab, were required for use in the fracture analysis. Before the bending moment arm stress was computed, the shear force P on the bolt was calculated from the DDAM-predicted shear stress τ_b using equation (1):

$$P = \frac{3}{4} A_t \, \tau_b = 17.354 \text{ kip}, \tag{1}$$

where A_t equals 0.764 in.^{2} and is the tensile area for a 1-1/8-inch-diameter UNC bolt. Note that the DDAM analysis originally used a 1-1/8-inch bolt and not the current 1.000-inch bolt. EB's decision to not rerun the DDAM solution for the smaller bolt was based on the following rationale: The shearing plane passed through the threaded region of the original 1-1/8-inch bolt; therefore, the shear force was calculated based on thread area equal to 0.764 in.^{2} The 1.000-inch bolt has a shorter thread region so that the shearing plane passes through the shank region and the shear force calculation is, therefore, based on shank area. The shank area of the 1.000-inch bolt, however, is 0.785 in.^{2} —only 2.80% greater than the threaded area of the 1-1/8-inch bolt. By using the threaded area of the larger 1-1/8-inch bolt, the computed shear forces would be slightly conservative.

A closed-form solution (equation (3))was established to predict the bending stress S_1 resulting from the eccentricity between the projector tab and ship foundation mid-planes. The effects of load transfer through friction generated between joint components were ignored because of the impulsive nature of the shock loading. All load transfer was assumed to be through bolt bearing. Bending was expected to occur only locally within the shank region since the shearing plane passed through the shank region; therefore, the shank area, rather than the tensile (threaded) area, was used to calculate the bending stress. The bolt bending moment arm L was assumed to equal the sum of the center G10 shim thickness t_{s_g10}, the CRES shim

thickness t_{s_cres}, one-half of the ship foundation thickness t_{found}, and one-half of the projector tab thickness t_{tab} as shown in equation (2), according to Bruhn:[6]

$$L = t_{s\_g10} + t_{s\_cres} + \frac{t_{found}}{2} + \frac{t_{tab}}{2} \quad = \quad 1.165\,\text{in.} \tag{2}$$

The maximum bending stress S_1 resulting from the shear eccentricity occurs on the surface of the bolt and is expressed by

$$S_1 = \frac{16\,P\,L}{\pi\,d_s^3} = 104.00\,\text{ksi}, \tag{3}$$

where d_s is the 1.000-inch shank diameter. (Note that NAVSEA concurred with the specific calculations and assumptions used to predict bolt bending stresses resulting from the shearing eccentricity.[7])

The original bolt torque requirement was 585 ± 59 ft-lb as specified by the ICD.[3] Assuming a torque coefficient κ of 0.2 and the upper limit of torque T, the corresponding pre-load force F_{pre} was 43.04 kip, as computed by Shigley:[8]

$$F_{pre} = \frac{T}{\kappa\,d_t}. \tag{4}$$

The threaded-region pre-load stress S_{pre} was 71.02 ksi. Because F_{pre} was constant between the threaded and shank regions and these regions have different cross-sectional areas, the pre-load stress in the shank region S_0 was 56.00 ksi.

The resulting shank bending stress S_1 was 104.00 ksi. The maximum combined tensile stress S_{total} (sum of S_0 and S_1) was 160.00 ksi, which exceeded S_{yield} by 39%. A resolution to the excessive shank axial stress was addressed by reducing F_{pre} to an acceptable level so that S_{total} did not exceed S_{yield}. Because of potential shear tear-out failures, no modifications to the projector tabs were permitted. Reducing S_0 from 56.00 ksi to 11.00 ksi netted the following results: (1) S_{total} equaled S_{yield}, (2) the reduced pre-load force was 8.64 kip, and (3) a factor of safety unity was achieved. Correspondingly, the reduced torque was 125 +0/-13 ft-lb. Concurrence by EB and the NUWC Division Newport was achieved, and the ICD was modified to reflect the reduced torque requirement to avoid overstressing the bolts.

4. NASGRO MODELING AND RESULTS

The projector assembly was categorized as a "shock grade-B" item mandating that fracture of the projector mounting bolts was not allowed for a single, 175-g port athwartship shock event. When the pre-load and bending stress components were determined and an initial flaw size was specified, a linear elastic fracture analysis was conducted.

The NASGRO fracture mechanics program was used to compute a mode-I SIF K_I for the solid cylinder/flaw geometry as shown in figure 4.

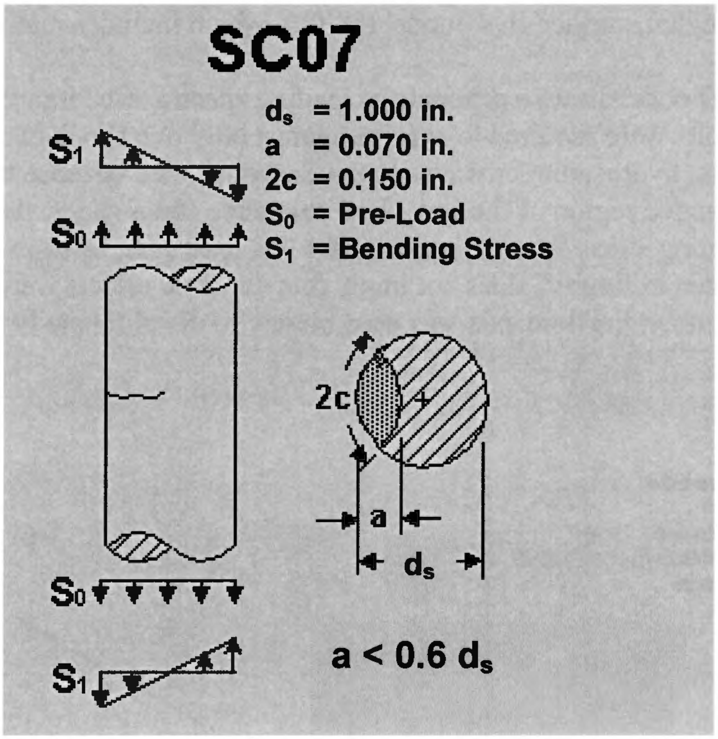

Figure 4. NASGRO Model SC07—Surface Flaw in Bolt Shank Region

This geometry reflects the shank region of the bolt having a 0.070-inch, thumb-nail surface flaw where the flaw depth a equals 0.070 inch and flaw width $2c$ equals 0.150 inch. These dimensions are related to the shank radius r_s by the following expression:

$$c = r_s \tan^{-1}\left[\frac{a(2r_s - a)}{2r_s(r_s - a)}\right]. \tag{5}$$

Equation (5) assumes that the crack front is circular and its intersection is always normal to the surface of the bolt.

The a/d_s ratio equals 0.070 and conforms to the condition that $a < 0.6d_s$, where d_s is the shank diameter. This method and the present flaw dimensions are consistent with the previously approved lightweight wide aperture array (LWWAA) fracture analysis by Northrop Grumman[9] except that the current focus is on a surface flaw located in the shank region rather than in the threaded region. The Northrop Grumman analysis used a predecessor of NASGRO, called NASA/FLAGRO,[10] to predict fatigue crack growth of Ti-6Al-4V ELI bolts. Per NAVSEA recommendation, since NASA/FLAGRO lacked an actual bolt thread SIF solution, Northrop Grumman used SIF solutions developed by Toribio et al.[11] Using the finite element method and some thread simplifications, Toribio explicitly modeled the effects of bolt threads but did not include mating threads from a nut or threaded hole. In the current fracture analysis of the HFSA projector bolts, SIF solutions were obtained for shank surface flaws using NASGRO's solid cylinder model (SC07); Toribio's method was not used. (As a point of interest, NASGRO 3.0 includes a threaded region, surface flaw model (SC08), which includes mating nut threads.)

The NASGRO code allows a schedule of loading spectra to be input; however, the titanium mounting bolts were required to survive against only one shock event. No other loads, such as hydrodynamic, hydrostatic, or wave slap, were considered because the projector is located in the free-flooded region of the sail. Two reference stress states, the total axial stress S_0, and the resulting bending stress S_1, were required for this load case, as shown in figure 5. The time dependence shown in figure 5 does not imply that dynamic effects were included in the NASGRO solution; rather, the time axis was used merely to discriminate between the axial and bending stress states.

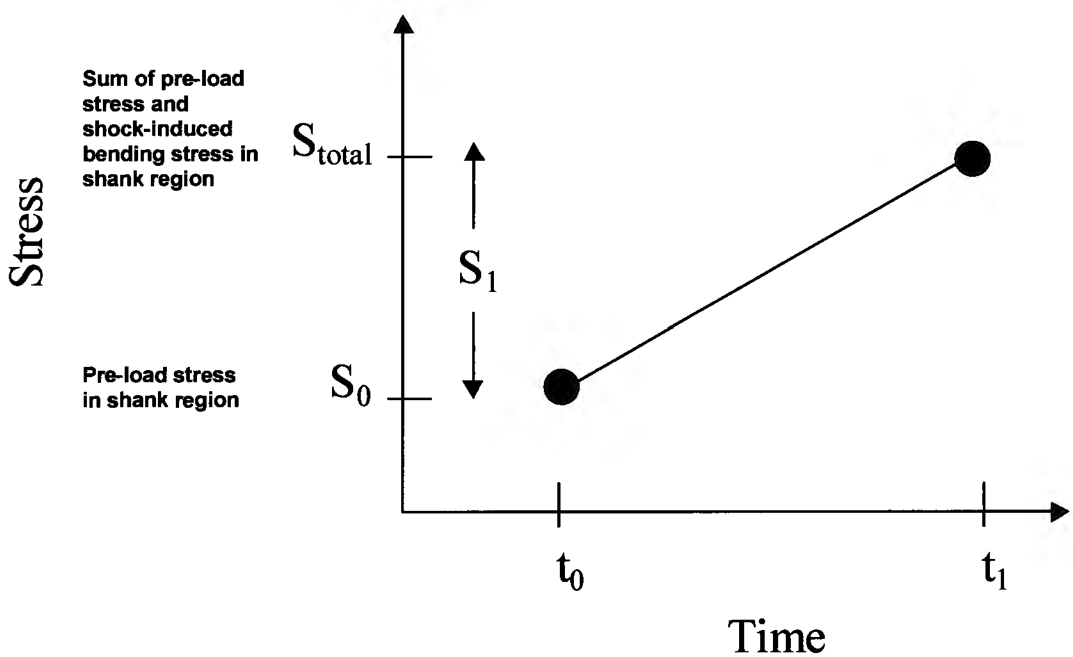

Figure 5. Axial and Bending Stress States

Forman and Shivakumar[12] derived the approximate SIF solution for the solid cylinder of figure 4 as

$$K_1 = \left(S_0 \ F_0(\lambda) + S_1 \ F_1(\lambda)\right)\sqrt{\pi \, a} \, , \tag{6}$$

where $F_0(\lambda)$ and $F_1(\lambda)$ are the magnification factors for the axial and bending stresses, respectively, and are defined by

$$F_0(\lambda) = g(\lambda)\left[0.752 + 2.02 \, \lambda + 0.371\left(1 - \sin\left(\frac{\pi \, \lambda}{2}\right)\right)^3\right], \tag{7}$$

$$F_1(\lambda) = g(\lambda)\left[0.923 + 0.199\left(1 - \sin\left(\frac{\pi \, \lambda}{2}\right)\right)^4\right], \tag{8}$$

where

$$g(\lambda) = 0.92 \, \frac{2}{\pi}\left[\frac{\tan\left(\frac{\pi \, \lambda}{2}\right)}{\frac{\pi \, \lambda}{2}}\right]^{\frac{1}{2}} \cos\left(\frac{\pi \, \lambda}{2}\right)^{-1}, \tag{9}$$

$$\lambda = \frac{a}{d_s} \, . \tag{10}$$

A plot of the nondimensionalized K_1 versus a/D is shown in figure 6.

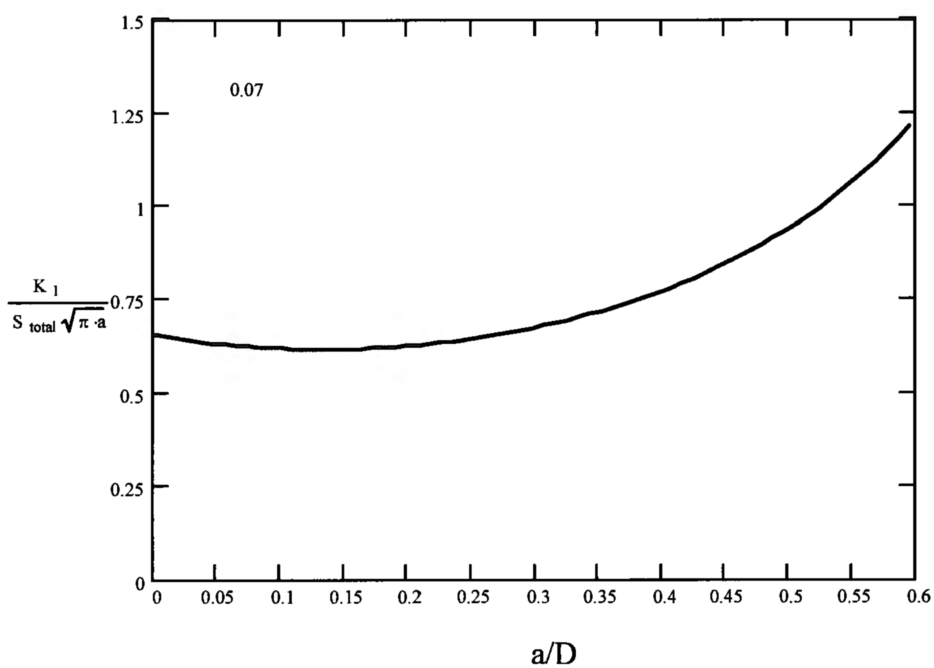

Figure 6. Nondimensionalized Stress Intensity Factor K_1 for Surface Flaw in Solid Cylinder—NASGRO Model SC07

Fracture mechanics results (see figures 7 and 8) were obtained for the 175-g port athwartship shock event, in which the titanium projector mounting bolts were pre-stressed to 11.00 ksi and subjected to a maximum shock-induced bending stress of 104.00 ksi. The corresponding maximum mode-I SIF K_I was 33.71 ksi-in.$^{1/2}$ and the critical flaw size a_{cr} was 0.2178 inch. In accordance with MIL-S-1222,[4] nondestructive testing is required to inspect the projector mounting bolts for defects using wet magnetic particle or liquid penetrant procedures. These procedures, described in SAE-J123,[13] can readily identify flaws of this size.

The fracture toughness of the titanium T7 bolt is specified by the ICD as 60.00 ksi-in.$^{1/2}$ To account for environmental effects, the fracture toughness was reduced to 70% or 42.00 ksi-in.$^{1/2}$ and is consistent with the LWWAA fracture toughness assessment. Using the environmentally reduced fracture toughness, the resulting critical flaw size a_{cr} was 0.0972 inch.

STRESS INTENSITY SOLUTION CHECK FOR SC07

U.S. customary units (inches, ksi, ksi sqrt(in.))

Shank Diameter d_s = 1.0000
Material Yield Stress = 115.00

S_0: Tensile Stress
S_0 = 11.00

S_1: Bending Stress
S_1 = 104.00

a	K_I	S_{total}	S_{total}/S_{yield}
0.0700	$\underbrace{33.7110}_{K_I}$	115.000	1.000

Figure 7. NASGRO Output of Stress Intensity Factor K_I for Shank Region Flaw

CRITICAL CRACK SIZE DETERMINATION

U.S. customary units (inches, ksi, ksi sqrt(in.))

MODEL: SC07

Shank Diameter d_s = 1.0000

K_{Ic} = 60.000
S_0 = 11.000
S_1 = 104.000

Iteration Table: (Newton's method)

Iteration No.	(a)initial	(a)final	(K_I)initial	(Residue)initial
1	0.0700	0.1868	0.3371 +02	-0.2629 +02
2	0.1868	0.2182	0.5478 +02	-0.5224 +01
3	0.2182	0.2178	0.6007 +02	0.6684 -01
4	0.2178	0.2178	0.6000 +02	0.1977 -04

Critical Crack Size:
a = 0.2178

Figure 8. NASGRO Output of Critical Crack Size for Shank Region Flaw

5. CONCLUSIONS

A CRES shim was added within the HFSA projector-to-ship foundation joint to facilitate alignment machining during installation. Subsequent stress analysis of the titanium projector mounting bolts indicated that the original torque requirements must be reduced to ensure survivability against the specified 175-g port athwartship shock event. The stress analysis was based on the maximum bolt shear stress from EB's DDAM results and a closed-form solution of the bolt bending stresses described in this report. A linear elastic fracture analysis conducted on the titanium projector mounting bolts using NASGRO predicted that (1) the critical flaw size was not reached, (2) crack growth would not occur for the one-time shock event, and (3) the fracture toughness of the re-crystallization-annealed T7 titanium alloy exceeded the maximum stress intensity factor by 16% and was sufficient even when reduced for environmental effects.

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